PEAK DEMAND REDUCTION WITH DUAL-SOURCE HEAT PUMPS USING MUNICIPAL WATER

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ABSTRACT

The objective of this project was to examine a dual-source (air-and/or water-coupled) heat pump concept which would reduce or eliminate the need for supplemental electrical resistance heating (strip heaters). The project examined two system options: switching on demand between completely air-source and completely water-coupled or using a concurrent partial water-coupled and partial air-coupled mode operation. The water supply for the water-coupled mode of operation would be the municipal water system. An estimate of the economic worth of this system concept was made by examining the incremental cost to install such a system against the expected savings associated with these systems.

INTRODUCTION

The trend in residential and light commercial heating system installations has been towards increased use of heat pumps in many parts of the United States. A problem has arisen since heat pumps have electric resistance heating elements (strip heaters) which are needed to provide sufficient heating capacity on the coldest days, and these strip heaters both cause increased electricity consumption by the consumer and cause the electric utility to increase its peak demands while reducing the load factor. The consumer ultimately has to pay for this increase in the utility's capacity since the base load is not increased at the same rate (due to the heat pump's COP).

This project investigated a concept which could help relieve this increased consumption and utility peaking situation. The concept includes developing a heat pump which is water-coupled only (1) when demanded by the electric utility or (2) whenever the strip heaters would have come on. The value of either development to the electric utility would be two-fold: first, the peak load could be reduced by demand-controlling; and second, the overall system KWH load would be basically maintained. The consumer would benefit both by reduced consumption and by electricity prices not having to be increased due to the peak loading. A further advantage of this dual-source concept is that it could be developed as a separate retrofit package as well as an integral part of new heat pump installations. Thus, the impact and benefit of such a development could be both broadened and hastened through retro-fitting existing air-source heat pumps.

Background

The use of water-source heat pumps has been a commercial reality for decades. Water from wells, ponds, rivers, lakes, etc., has been supplied to the heat pump in a "flow through once" (open system) mode. There have also been extensive investigations into closed loop ground-coupled water source heat pump systems, both in the past [1-4] and recently [5-7]. However, previous investigations involving the dual air-water source concept are very limited, to the best of the authors' knowledge.

One of the current authors examined system configurations involving the dual-source concept for peak demand reduction [8], but the analysis was limited to switching between the air and ground-coupled water sources, not using them concurrently. A second study [9] examined both concurrent and switching modes of operating with the air and water sources, but used small ground-coupled water loops. An experimental program is underway in Louisiana as a joint Louisiana State University/Louisiana Electric Distribution Cooperative/Heat Pump Company project, with the objective "...to develop, evaluate and field test new electric heat pumps that utilize both air and water source simultaneously..." [10]. The LSU program scope includes operating a dual-source heat pump with the water coming from water-coupled loops (horizontal and vertical closed systems) and wells or city supply systems (open systems) with water disposal after use.

Objective

This study examined the concept of using municipal water as the heat source for dual-source heat pumps in place of strip heaters. The water source would be controlled to either (1) supply all the heat required by the heat pump for short periods during the utility peak periods or (2) supply supplemental heat in place of the strip heaters during the entire period when the air-source alone cannot meet the load, thus reducing the overall electric demand. The investigation of the two options was accomplished using two broad task areas:

- Determination of the technical performance and water requirements for each operational mode, and
- Evaluation of the options' economic feasibility, using cost and savings relative to the value of the reduced peak demand and electricity consumption.

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SYSTEM CONFIGURATION

The part which must be "added on" to the conventional air-source heat pump consists of a water-to-refrigerant (water source) evaporator heat exchanger. The water-to-refrigerant evaporator heat exchanger can be installed in either parallel or series with the air-to-refrigerant (air-source) evaporator. Parallel installation would most likely require a switching valve for refrigerant flow control. The choice of series connected air- and water-source evaporators has been made for the LSU project, with the heat pump component layout as shown in Figure 1 [10].

Of technical concern with a series versus parallel installation of the water-to-refrigerant heat exchanger is the problem of water freezing inside the exchanger. During steady-state operation of the exchanger, the temperature drop of the incoming water would be controlled (or designed) to prevent freezing. However, if water remained in the exchanger during the air-source mode of operation, a series-connected exchanger would continue to flow refrigerant and would ensure freeze-up. In fact, a series-connected exchanger would probably cause freezing of the incoming water during start-up of the water-source mode if the refrigerant had been circulating through the exchanger. A parallel-connected exchanger would be more likely to avoid these problems, but would involve more hardware.

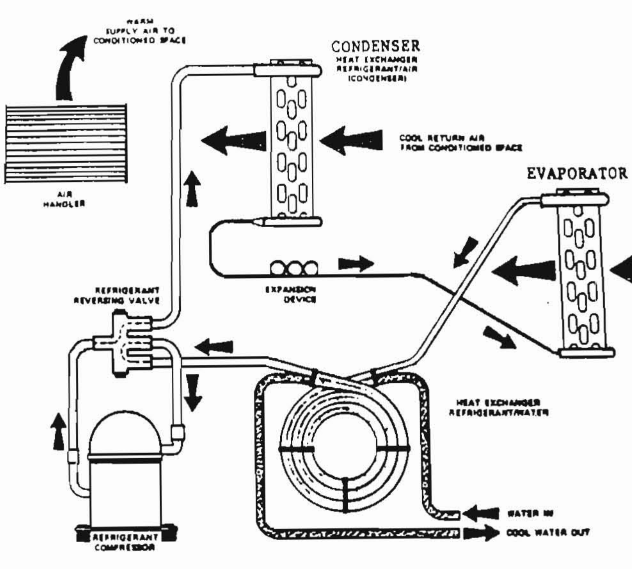

FIGURE 1. Dual-Source Heat Pump Schematic Illustrating Water-to-Refrigerant Heat Exchanger Connection [10].

OPERATING MODES

This study examined two modes in which the dual-source heat pump could be operated (1) demand-controlled mode and (2) supplemental mode.

<u>Demand-controlled Mode</u>

In this mode of operation, the heat pump uses either the water or the air as a heat source, but not both concurrently. The water source is only operated when the local utility is in a peak demand period and wants to control (reduce) demand. Between these periods of peak demand control, the air-source coupled system would be operating conventionally, including using strip heaters as needed. Thus, the amount and frequency of the water demand, i.e. how often the water source is used, is determined by the peak demand and off-peak periods themselves. The utility would control the heat pump water mode using a "dispatch-type" load control system. Knowledge of the expected periods is required for system analysis.

This information concerning when and how long demand controlling is desired and the time between these periods was obtained from the Central Electric Power Cooperative which distributes the power and provides demand control for electric coops in two-thirds of South Carolina [11]. Table 1 presents this information in terms of the expected system operating times which corresponded to the peak demand control times for three years' heating seasons. To read this table, one would look under "First Year" and see that the system would be operated for 3 hours (Demand Period column) followed by 93 hours of non-operation (Off-Demand Period column), then 2 hours of operation (Demand Period) followed by 48 hours of non-operation, etc. It was fortunate that these three years represent a broad range of expected demand peaks and controlling, with "First Year" (1985-86) being a very moderate heating season, "Second Year" (1986-87) being a fairly nominal heating season, and "Third Year" (1987-88) representing a very heavily peaked and controlled heating season for South Carolina.

<u>Supplemental Mode</u>

In the supplemental mode of operation, the water source supplies supplemental heat in addition to that supplied from the air-source. The water source heat supply would be equivalent to that supplied by strip heaters in a conventional air-source heat pump. The supplemental mode of operation would be initiated when the air-source capacity drops below the heat load (balance point). Figure 2 represents an example air-source heat pump capacity and house heating load, both as a function of ambient temperature. The effect of the supplemental water source is to increase the heat pump capacity by increasing the "ambient temperature the heat pump sees." The amount of this supplemental water-source heat supplied is given by the difference between the load curve and air-source heat capacity curve at a given ambient temperature below the balance point.

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TABLE 1. Demand-Controlled System Operating (Demand) and Non-operating (Off-Demand) Periods for Three Heating Seasons in South Carolina.

FIRST YEAR		SECOND YEAR		THIRD YEAR	
Demand/Off-Demand (HOURS)		Demand/Off-Demand (HOURS)		Demand/Off-Demand (HOURS)	
3.0	93.0	2.0	142.5	3.25	71.0
2.0	48.0	1.0	261.5	3.0	20.0
2.5	91.5	3.5	22.5	2.5	236.0
5.0	331.5	2.5	40.0	4.0	20.5
3.5	440.5	2.0	10.0	4.0	284.0
5.0	7.0	5.0	18.5	2.0	21.0
5.0		3.0	21.0	5.0	91.0
		4.5	260.0	4.0	20.0
		3.0	20.5	4.5	19.5
		4.5	19.5	5.0	7.0
		3.25	129.0	3.75	8.0
		3.0	9.0	16.0	8.0
		3.5	9.0	6.75	90.0
		3.5	394.0	2.75	46.5
		1.5	106.0	2.5	20.0
		1.75	9.5	3.0	23.0
		3.0	45.0	2.0	260.0
		3.25		3.75	20.0
				3.75	20.5
				0.75	261.0
				3.75	

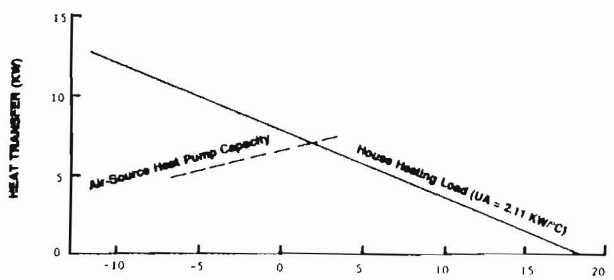

FIGURE 2. Example Air-Source Heat Pump Capacity Curve and House Heating Load as a Function of Ambient Temperature.

It should also be noted that the use of a supplemental mode (air-and-water sources operating simultaneously) evaporator has not been invented. Unless the two sources were at the same temperature, the higher temperature source would cause the evaporator temperature to rise above the lower source temperature, eliminating it as a source. Thus, this study views the investigation of the supplemental mode as a "thought experiment" to determine whether such a device would even be worth inventing.

TECHNICAL ANALYSES

The technical analyses of the two operational modes was directed towards determining the heat pump system total water supply requirements, the peak electrical power requirements, and the total electrical energy requirements for a heating season in Charleston, SC. The choice of Charleston was based on weather data availability for a South Carolina city and the fact that Charleston's climate is fairly representative of much of the Southeastern United States inland to the fall-line.

System Description

In order to examine system performance, two dual-source systems (one demand-controlled, the other supplemental mode) and a conventional air-source heat pump with strip heaters were postulated. All the heat pump systems delivered 10.5 KW of heating to a residence at the peak demand period; this corresponds closely with the design-day load but is a bit more conservative (see Figure 2: House Heating Load Curve).

The heat pump COP was assumed as 2.5 operating with the water source, regardless of the ambient temperature. The air-source heat pumping was assumed to have a COP = 2.0 and could supply one-half the load on the coldest days (peak demand periods), both for conventional air-source heat pump and supplemental mode operation. This information corresponds to the Figure 2 load and capacity curves.

Demand-Controlled System

As described above, the demand-controlled system switches from completely air-source to completely water-source heat pumping. The number of hours the system is required to be on water-source only depends on the utility's demand, and the sum from the schedules for three different years (Table 1) gave 26.0, 53.75 and 86.0 hours respectively for the lightest to the heaviest heating season.

To calculate the water required for demand-controlled operation, the heat removal rate from the water is found using the COP definition which leads to:

$$\begin{aligned} Q_{L,ws} &= Q_{H,ws}(1 - 1/COP_{ws}) \\ &= 10.5(1 - 1/2.5) \\ &= 6.3 \text{ KW} \end{aligned} \quad (1)$$

where the subscript "ws" refers to water-source operation. Assuming a 10°C temperature drop in the water as it flows through the water-to-refrigerant exchanger, the required flow rate of water is:

$$\begin{aligned} m &= Q_{L,ws}/c\Delta T \\ &= 6.3/(4.19)(10) \\ &= 0.15 \text{ Kg/sec} \end{aligned} \quad (2)$$

and the total seasonal water use is found by multiplying by the hours per season from above. The results are presented in Figure 3 as the demand-controlled system water requirements

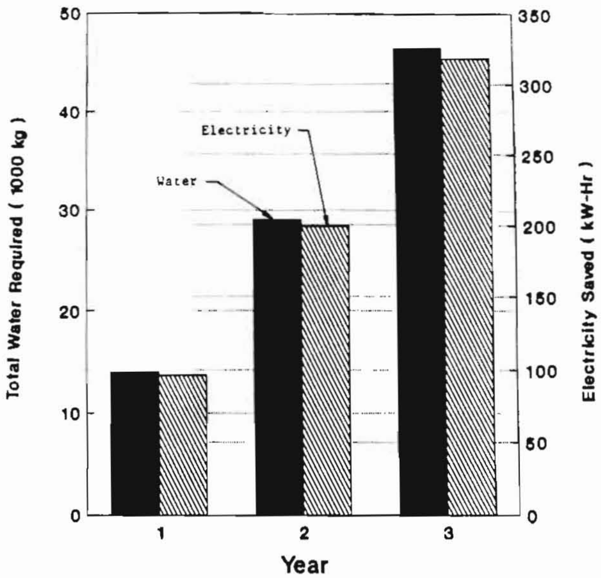

FIGURE 3. Demand-Controlled Water Requirements and Electricity Saved Versus the Conventional Heat Pump.

for each of the three years, with 29,000 Kg of water needed for operation in a nominal (Year #2) year.

The peak electrical requirement for the demand-controlled system would be the power to run the compressor during fully water-sourced operation:

$$W_{COMP,WS} = Q_{H,WS}/COP_{WS} = 10.5/2.5 = 4.2 \text{ KW} \quad (3)$$

In comparison, the conventional heat pump would be supplying one-half the heat from the air-source with:

$$W_{COMP,AS} = 5.25/2.0 = 2.63 \text{ KW} \quad (4)$$

and the other half would be supplied by the strip-heaters (5.25 KW electric). Table 2 summarizes these results and illustrates the fact that the demand-controlled system has a peak power requirement 3.7 KW lower than the conventional heat pump. Multiplying this 3.7 KW reduction by the number of hours of water-source operation for each of the three years provides the amount of electricity saved per year relative to the conventional heat pump system. These yearly electricity savings are presented in Figure 3, with 199 KWHR saved during the nominal year.

Supplemental System

In the supplemental mode of operation, the water-source supplies the heat which would have been supplied by the strip-heaters. Quantitatively, the water-source supplied heat is a function of the ambient temperature and is the difference between the house heating load and the air-source capacity curve at a given temperature (see Figure 2). Thus, the rate of heat removal from and the required flow rate of the water would be calculated using Equations 1 and 2 as before, but with a $Q_{H,WS}$ determined as a function of ambient temperature. Table 3 presents the bin-method weather data for Charleston, SC [12] with the supplemental water-source rate of heat supplied to the house and the rate of heat removed from the water as a function of ambient temperature. The annual water requirements for the supplemental mode system are presented in Figure 4 as a function of ambient temperature, again assuming a 10°C temperature drop in the water source heat removal process and using the number of hours at each bin temperature. The total seasonal water required for operating the supplemental mode system in Charleston, SC is 19,900 Kg of water.

The peak power required for the supplemental mode system is determined when the system is supplying a heating load of 10.5 KW. One half the load is met from the air source with a COP

TABLE 2. Peak Electrical Power (KW) Requirements and Differences for the Two Dual-Source Systems Versus the Conventional Heat Pump

	Conventional Heat Pump	Demand-Controlled HP System	Supplemental HP System
Air-Source:	2.63kw	0.0	2.63
Water-Source:	0.0	4.2	2.10
Direct Electric:	5.25	0.0	0.0
	7.88kw	4.2kw	4.73kw
Peak Power Reduction:	0.0 (basis)	3.7kw	3.2kw

TABLE 3. Bin Data and Calculated Heat Requirements for Supplemental Mode Operation in Charleston, SC

Bin Temperature (°F)	Number of Hours per Season (hours)	Supplemental Heat Supply Rate (kW)	Heat Removal Rate from water (kW)
35 - 39	352	0.0	0.0
30 - 34	225	1.218	0.731
25 - 29	96	3.138	1.882
20 - 24	30	5.058	3.035
15 - 19	6	6.978	4.187
10 - 14	0	8.898	5.339

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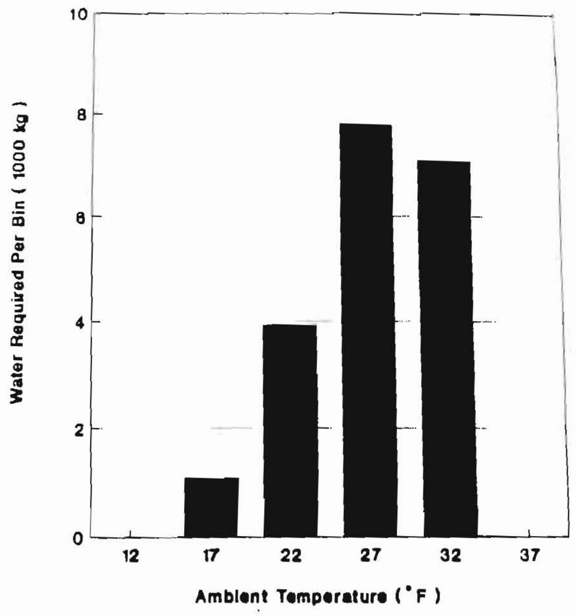

FIGURE 4. Water Required for Supplemental Mode as Function of Ambient Temperature in Charleston, SC.

= 2.0, and the other half of the load comes from the water source with a COP = 2.5. Using Equation 3 with these values results in the values presented in the last column of Table 2, giving a peak power reduction of 3.2 KW for the supplemental mode system relative to the conventional heat pump.

The electricity saved relative to the conventional heat pump is equal to the heat removed from the water source. In the conventional heat pump, strip heaters would have supplied this energy. In calculating the annual savings, the rate of heat removed and the number of hours for a given bin were multiplied, and the sum of the four bins represents the annual electricity savings (461.3 KWHR).

SYSTEM ECONOMICS

A complete system economic optimization is not appropriate at this stage of concept development. However, an overall examination of the costs and benefits of these two systems is valuable in determining general feasibility of the concepts.

Costs

Several manufacturers of water-to-refrigerant heat exchangers were contacted to determine the price range for a 10.5 KW exchanger, and prices ranged from $87 to $272 for the bare heat exchanger. The labor to install this exchanger on a heat pump (retro-fit) was estimated at 7 man-hours at $21/hour ($147). It is also necessary to have a solenoid valve which will control (on/off) the water supply. Such a valve would be activated by the utility (demand-controlled mode) or by the strip-heater activator in a thermostat. The cost of such a valve installed is estimated at $150. The system equipment cost is the sum of the above costs. Thus, it was assumed for the analysis below that the water-source add-on system costs $500. Again, it is realized that no two-simultaneous heat source evaporator is available, but it will be assumed to "cost" $500 more than a single-source evaporator.

The cost of the municipal water supplied will differ for the two systems. By contacting various municipalities in South Carolina, water costs varied, but hovered around $0.25/10^3$Kg. With the heating season water requirements determined earlier, the water for the nominal year demand-controlled heat pump costs about $7 and the supplemental heat pump water costs $5.

Savings

The total savings with the water-coupled systems are due to savings in two areas: peak power reduction and reduced energy consumption. Both of these savings are relative to the power used by the conventional air-source heat pump.

The peak power requirements for the water-coupled heat pumps and the air-source heat pump and their differences are shown in Table 2. The benefit to the utility of reducing each KW of peak power is $6/month for each of the six months in the heating season [11]. Thus, from the previously determined values in Table 2, the $36/KW benefit translates to a saving of $133 for the demand-controlled system and $115 for the supplemental system on an annual basis. It is assumed in this study that these savings would be passed to the consumer from the utility via an appropriate "rebate" when the system was installed or connected to the control system.

The electrical energy saved due to lower power consumption by the demand-controlled mode unit is calculated using the nominal (second) year (see Table 1 and Figure 3). Assuming electricity costs 6 cents/KWHR [11], the annual energy savings is $12. Thus, the total of the peak and consumption savings is approximately $145 per year for the demand-controlled mode system. Subtracting the $7 annual water cost gives a net yearly savings of $138.

The electrical energy saved due to lower power consumption by the supplemental mode unit was calculated using the bin method for Charleston, SC [12], using the difference between the supplemental-mode energy consumption and what the strip heaters would have used in the hours below the balance point. With 6 cents/KWHR, the annual energy savings is $27 based on saving 461.3 KWHR. The total peak and consumption savings for the supplemental mode system is $142 minus the $5 annual water cost for a net yearly savings of $137.

Payback

Simple payback is used due to the basic assumptions being used for costs and savings at this initial level of economic analysis. The simple payback is defined as:

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$$\text{SIMPLE PAYBACK} = \frac{\text{TOTAL SYSTEM COST}}{\text{NET YEARLY SAVINGS}}$$

and for the two system types with costs of $500 and annual savings of approximately $138:

SUPPLEMENTAL SYSTEM PAYBACK

OR

DEMAND-CONTROLLED SYSTEM PAYBACK = $500/$138/YR
= 3.6 YEARS.

Thus, for these assumed off-the-shelf items and proto-typical installation costs, the dual-source systems would take approximately 3.6 years to payback their initial costs via yearly savings. Normally, consumers look for a 3-5 year payback on this type of capital investment, so system costs appear to be appropriate to ensure substantial consumer involvement.

CONCLUSIONS

The water-source "add-on" for air-source heat pump applications appears to be technically feasible for the demand-controlled mode dual-source heat pump systems. The system economics are attractive for a system at the developmental level.

Summary of Results

The annual performance characteristics of a dual-source heat pump operating with a municipal water supply was determined for two modes of operation, with annual water requirements on the order of 2,500 Kg per KW of installed heat pump capacity. A preliminary estimate of economic worth of these conceptual systems was conducted by examining both the cost to install such a system and the yearly savings associated with that system. A simple payback of three to four years was found for both the demand-controlled unit and for the supplemental mode unit.

Future Activities

In this preliminary examination, several technical and economic issues were not addressed which need further resolution in any future development of this water-source concept.

On the technical issues, the insertion of the water-refrigerant heat exchanger in series or in parallel with the heat pump evaporator is an undetermined matter. The issue of freezing the water in the exchanger appears to be the major decision factor, but volume/flow factors in the evaporator, and dual-evaporation sites are technical issues which need to be addressed.

The technical and economic feasibility for retro-fit also needs further examination. The disposal of the water after use needs to be examined in light of water conservation, even though the overall usage would be fairly small. The use of these system concepts with larger systems and/or in areas with longer heating seasons would lead to more significant water usage.

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